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Investigate Advanced Heat Exchanger Designs
of Compact Heat Exchangers When Operating
In a Marine Environment

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Prepared By
The Air Preheater Co., Inc.
Wellsville, New York, 14895

Signed: Henry N. Osborn

Henry H. Osborn, Manager of
Applied Research Department

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INVESTIGATE ADVANCED HEAT EXCHANGER DESIGNS OF COMPACT HEAT EXCHANGERS WHEN OPERATING IN MARINE ENVIRONMENT

This is the second of the Quarterly Progress Reports under this contract covering the period from September 26, 1965 to December 31, 1965. The contract was received by the Air Preheater Company, Inc., effective June 24, 1965, and will terminate December 24, 1966, covering a period of 18 months. The Intertech Corporation of Princeton, New Jersey, has been issued a subcontract for various analytical and theoretical studies pertinent to the scope of the contract.

A. Objectives of the Reporting Period

Briefly, the objectives of the reporting period of this quarterly report were:

1). to complete the installation of the new low turbulence wind tunnel, perform flow calibration, determine the tunnel turbulence level, retest previously tested surfaces, and start testing new perforated surfaces.

2). to continue the literature search, make analytical studies of the effect of variation in matrix inlet temperature on the maximum slope calculation procedure, and to establish a procedure for running tests that will optimize the number of experiments needed for correlating the desired number of parameters.

3). to construct a flow visualization tunnel and commence tests to study the flow behavior over perforated plates.

4). to continue bi-weekly reports as per schedule.

B. Work Accomplished

1). The low turbulence wind tunnel has been received from Intertech Corporation, Princeton, N.J. It has been assembled and instrumented by Air Preheater Company personnel. The description of the tunnel and components is given in Appendix B1.

The flow calibration of the orifices in the tunnel has been satisfactorily completed. The quasi-steady state pressure drop tests were run on a matrix of 28 gage flat plate, with 1/16" spacing. For this matrix the fanning friction factors calculated as a function of Reynolds' number were in oxcellent agreement with the theoretical values for a rectangular passage of the same aspect ratio and L/D_h ratio. As a result of these tests, the pressure instrumentation and the air flow measurement technique were established as correct.

Heat transfer testing was started on the same core but several problems with the tunnel components and the temperature instrumentation have slowed progress and to date, the test results are not up to expectations.

2). A method of performing the tests, which will optimize the number of tests required to correlate the desired parameters, has been chosen. The previous test agenda included too many small variations and an excessive number of points for each run, making the testing very time-consuming. Individual core boxes have been prepared for all the surfaces to be tested. This makes it possible to quickly run a few points for each core. After reviewing these results, more extensive testing can then be done on cores that will yield significant results. Also, this method will facilitate exploring the effect of turbulence generation methods on the different matrices and also facilitate making check runs or additional tests at later dates in the program.

3). To date, the analytical efforts have been directed toward determining the effect on heat transfer of a varying inlet temperature to the test matrix at the start of the blowdown period of testing. A summary of this work is presented in Appendix B2.

4). Flow visulatization methods were investigated and a test arrangement fabricated. The design considerations were: (1) to provide information on flow over various slotted and perforated plates and, (2) to indicate the heat transfer characteristics of scaled-up versions of these plates. The details are given in Appendix B3. Experimental investigations with the flow visulization setup will be started in the near future.

5). The means of artificially producing disturbances were established. A rotating pin wheel device has been designed to create artificial turbulences. Also, a crossed-bar screen matrix will be used as an alternate method for generating turbulence in the tunnel. The details of these devices are described in Appendix B4.

6). The literature survey was continued, and a Bibliography is being prepared under the following 14 headings:

1. General
2. Single-Blow Technique
3. Regenerative Heat Exchanger
4. Porous Matrix Flow
5. Turbulence Measurement
6. Flow Pulsation
7. Flow Vibration
8. Plate Heat Transfer
9. Perforated Plate Flow
10. Scaling of Flow
11. Flow Visualization
12. Design of Experiment
13. Testing Facilities
14. Heat Transfer Surfaces

A list of 74 items collected during the reporting period, are given in Appendix B5.

C. Conclusions

The new wind tunnel has been installed and the flow calibration of the orifices completed. Several problems with the temperature instrumentation have prevented any acceptable heat transfer tests to date. It is expected that by using more efficient methods for testing (the use of separate core boxes for each surface which will allow abbreviated testing with more complete testing of selected surfaces later), and by accelerated testing, the wind tunnel program will soon be back on schedule.

D. Projected Progress During The Next Reporting Period

1). the heat transfer test procedure will be firmly established.

2). commence testing - Phase I of the program; testing of previously evaluated perforated surfaces with minimum flow turbulence.

3). commence testing - Phase 2 of the program; determine the effect of flow disturbances on flat plate performance.

4). commence testing - Phase 3 of the program; testing of other perforated surfaces with minimum flow turbulence.

5). continue the technical literature survey pertinent to the proposed scope of work.

6). commence flow visualization experiements.

7). continue analytical treatment of associated problems with single blow test technique.

8). continue Bi-Weekly progress reports as per schedule.

Appendix B1

DESCRIPTION OF TEST APPARATUS

The heat transfer and pressure drop tests are conducted in a low turbulence wind tunnel having a 6 inch square test section. An induced draft fan driven by a 15 HP, DC motor supplies air to the tunnel. Instrumentation is provided for metering the air, for measuring the air temperature before and after the test core, and for measuring the static and differential pressure through the core.

Low Turbulence Wind Tunnel

The low turbulence tunnel was designed and manufactured by the INTERTECH CORPORATION. The tunnel incorporates three very important principles necessary to achieve the low turbulence level at the test section. (a) The tunnel operates on the suction principle, i.e., air is drawn by a motor-operated fan through the inlet and consequently through the test section and discharged by the fan at the end of the tunnel. (b) At the inlet section, a generously sized settling chamber together with a well designed entrance cone permits the air to be accelerated with a minimum velocity variation at the test section. (c) A series of special steel screens in the settling chamber will dampen out turbulence.

The wind tunnel is depicted in Figure 1. The overall dimensions of the tunnel are: 22 feet long, 10 feet high and 10 feet wide. There are four parts of the tunnel, viz., (1) the inlet section consisting of a settling chamber and an accelerating cone, (2) the test section, (3) the diffuser section and (4) the turbo compressor.

A description of each tunnel part is as follows:

(1) There are four stainless steel screen panels in the settling chamber. Each screen is tightened independently in the horizontal as well as in the vertical axes. In this fashion possible vibration of the screen, which can lead to high turbulence levels in the tunnel, is prevented. The inlet scroll to the settling chamber has a sufficiently large radius (four inches) to avoid flow separation and disturbances at the inlet to the settling chamber.

(2) The test section consists of a test core, three thermocouple grids, two piezometer rings, and a heating grid. The test section has a 6" x 6" cross section and is about 5 ft. long, made from 3/16" thick hot rolled steel plates and insulated on the interior by 1/2" thick plywood sheet.

(3) The diffuser section, located after the test section, has an area ratio of 2 over a length of 2 feet. This makes possible the proper operation of the tunnel at higher air flows than anticipated in this project. The orifice plate for metering the air is located at the beginning of the diffuser section.

(4) The air is drawn into the tunnel by a turbo compressor (rated at 2000 cfm at 30 in. water static pressure). The fan has 12 radial vanes. It is driven by a 15 HP, 3500 rpm, 550 volt compound wound motor with a DC, SCR continuous control, which provides a speed range of 500 to 3500 rpm. The power supply is 440 V, 3 phase, 60 cps.

Great care was taken to avoid any generation of turbulence in the wind tunnel. All inside weld seams were smoothly ground and all mating parts were deburred. The inside surfaces of the test section and insulation were sanded smooth. Special design considerations for minimizing turbulence were taken into account in the fabrication of the heating screen and the thermocouple grids located upstream of the test core. Smooth transition ducts were provided before and after the test section to avoid any interruption in the air flow.

Air System

The air flow is primarily controlled by changing the power input to the fan motor. At very low flow rates, corresponding to motor speeds below 300 RPM, the air flow is controlled by a manually operated blast gate in the exhaust duct.

Atmospheric air enters the inlet section and passes through four dampening screens. The dampening screens reduce the intensity of the free stream turbulence. The air then enters the accelerating cone which has a contraction area ratio of 114 to 1. This permits a greater decay of turbulence in a given length of settling chamber. The large contraction ratio also provides for a low air velocity in the settling chamber for the desired range of velocity in the test section, thereby insuring a low pressure drop for the screens.

The air then passes through the test core, a filler box, a diffuser cone, and finally, the radial vane fan. The test core is 12" long. A piezometer ring is located at the end of, the 30" filler box. The transition section connected at the piezometer ring box converts from the 6" x 6" square section to a 8-1/2" round section. The orifice plate is located at the end of this transition piece and at the beginning of the diffusion cone. Three orifice plates, 6", 2-1/2" and 1-1/4" in diameter, are used to cover a range of air flows from 100 lbs/hr to about 8000 lbs/hr. After leaving the fan the air is exhausted through a 12" duct.

Pressure Instrumentation

Pressure instrumentation is provided for measuring the test core pressure drop, the pressure drop across the orifice plate, pressures upstream of test core, and pressures upstream of the orifice plates. Well-type single leg water manometers, high precision inclined draft gages, and a precision micromanometer are used to measure these pressures.

Piezometer rings are used at the gage pressure and pressure drop stations. One piezometer ring is located in the test section upstream of the test core, the other is located about 32" downstream of the test core to assure complete pressure recovery from the core. Each of these rings has a total of 44 holes of 0.04 inch diameter spaced around the inside tunnel wall at intervals of 1/2 inch. Due to the large number of holes in these rings, a very fast response is achieved. The L/d ratio of these holes is 4.7 which meets with the ASME recommended value of 2.5 or higher. A comparatively large value of this L/d ratio is preferred so that small fluctuations in pressure will be dampened out.

Four standard pipe taps are evenly spaced around each side of the orifice and connected together with 3/16" diameter steel tubing. The pipe tap center line is located 1 inch from the orifice plate face. The diameter of the pipe tap hole is .042 inch, creating an L/d ratio of 4.7.

Copper tubing is used to connect the piezometer rings and pipe taps to the manometers.

Temperature Instrumentation

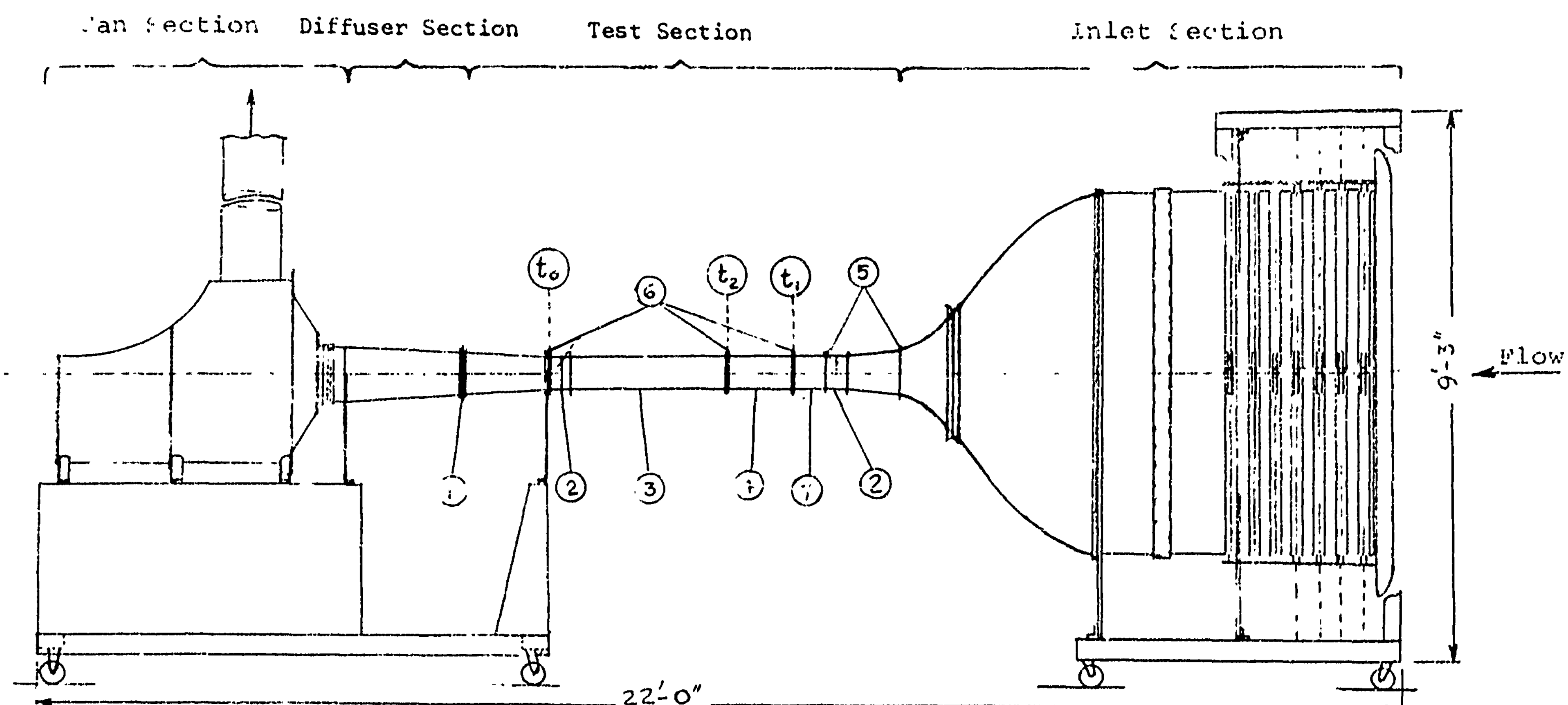
Temperature instrumentation is provided for measuring the temperature ahead of the orifice, the temperature upstream and downstream of the heating screen, and the temperature after the test core. All steady state temperatures are measured with a grid of ten 30-gage iron-constantan thermocouples. Transient temperatures are measured with a grid of ten 40-gage iron-constantan thermocouples. Ice baths are used for reference junctions.

The thermocouple grids have been constructed so that they introduce a minimum turbulence in the free stream. Bare thermocouple wires are used and stretched across the section. The thermocouple wires from the hot and cold junctions are connected in series at a terminal block from which the two signal leads go to an x-y recorder.

Heat Source

In the single blow technique, the air passing through the test matrix is first preheated by an electrically heated screen upstream of the core. The screen is made up of one layer of 100 mesh, 0.0031" diameter Nichrome-V wires. The Nichrome screen has a high resistivity, a low thermal conductivity and a low specific heat. Because of this, the screen has a very fast response time to input power variations.

The size of the screen is 8" x 6". The screen is sandwiched between two 1/2" x 1" x 13" copper bus bars and held firmly there by screws and brazing. The screen and bus bar assembly is mounted in a 1" thick plywood frame which is inserted into the tunnel for heat transfer testing. The power to the heating screen is supplied by a variable 600 amp DC welder.



(1) Orifice Plate

(2) Piezometer Ring

(3) Filler Box

(4) Core Box
6" x 6" x 12'

(5) Turbulence Generating Device

(6) Thermocouple Grids

(7) Heating Grid or
Turbulence Measurement Section

Figure 1. Low Turbulence Wind Tunnel

Appendix B2

SUMMARY OF ANALYTICAL WORK

To date, the analytical efforts have been directed toward considering certain assumptions made in the maximum slope calculation procedure. Principally, the concern has been with the effect that variations in the matrix inlet fluid temperature have on the method.

In the single blow technique as used in this study, the matrix is preceded in the flow direction by a grid of ten thermocouples connected in series and a fine mesh screen which can be heated electrically. In operation, current is supplied to the heating screen for a period of time sufficient to allow the matrix to reach a uniform temperature. At this time, the heating current is turned off and the difference between the matrix inlet and outlet temperature is recorded. To use the single blow technique properly, it is important that the resulting inlet temperature closely approximates a step function. The actual behavior of the inlet temperature distribution is therefore of interest.

The principal cause for a departure from a step change in the inlet temperature is the finite heat capacity of the screen material which causes the screen to release heat slowly to the fluid after the heating current has been turned off. The details of the process involved have been explored and the results will be presented here. The cooling process of the screen has been found experimentally (with good theoretical justification) to be well represented by a simple exponential. The temperature distribution in the fluid leaving the screen can thus be written as

$$t_2-t_1 = (t_{20}-t_1)\, e^{-\theta/\theta_s} \tag{1}$$

where t_1 and t_2 are the temperatures of the fluid approaching and leaving the screen respectively, t_{20} is the temperature leaving the screen while the latter is being heated, θ is the time measured from the instant the cooling of the screen begins, and θ_s is a time constant for the cooling process.

The time constant θ_s for the screen can be found for the special case of infinite thermal conductivity in longitudinal direction as

$$\theta_s = \frac{(Wc_p)_s}{(\dot{w}c_p)_f} \cdot \frac{1}{1-e^{-N_{tu,s}}}$$

from Mondt's analysis (1). $(Wc_p)_s$ refers to heat capacity of screen, $(\dot{w}c_p)_f$ refers to fluid heat capacity rate, and $N_{tu,s}$ refers to the number of heat transfer units of the screen.

The temperature distribution is measured by a grid of thermocouples which themselves can be characterized by a time constant θ_{tc}* related to an exponential decay to a step in fluid temperature. The temperature indicated by the thermocouple grid when subjected to the input given by equation 1 has been shown to be represented by the equation (if $\theta_{tc} \neq \theta_s$),

$$t_{tc}-t_1 = (t_{20}-t_1)\, \frac{e^{-\theta/\theta_s} - \frac{\theta_{tc}}{\theta_s} e^{-\theta/\theta_{tc}}}{1 - \frac{\theta_{tc}}{\theta_s}} \tag{2}$$

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* θ_{tc} is defined as $\left(\frac{Wc_p}{hA}\right)_{tc}$ where each of the quantities in the bracket refers to the thermocouple grid.

with θ now measured from the instant the temperature wave reaches the thermocouple grid.

Equation 2, using a variation of the maximum slope method, has been used to determine the time constants of both the thermocouple grid and the heating screen. This method uses the dimensionless temperature at which the maximum slope occurs to determine the ratio of time constants θ_{tc} / θ_s from which, with the value of the maximum slope itself, both time constants are determined. A typical cooling curve of the screen is shown in Figure 2, and the results of two determinations are given in Table I. It is interesting to note that the ratio of the time constants is independent of low rate and that, within 10 percent, either time constant decreases approximately as the inverse square root of the flow rate (Reynolds No.). If the thermocouple time constant is neglected in the screen time constant, the resulting screen time constant would be in error by as much as 50 percent.

Table I
Heating Screen and Thermocouple Time Constants

Flow Rate (lbm/hr)	θ_s (sec)	θ_{tc} (sec)
390	0.67	0.2
3790	0.25	0.076

Since the inlet temperature profile to the matrix is a decaying exponential, as given by Equation 1, we can proceed to consider its effect on the outlet temperature profile and the use of the single blow technique. Although the ultimate resolution of this problem will probably require numerical solution, the problem can be formulated and, in certain cases, closed from solutions developed. This analysis has been carried out in some detail and will be continued during the next quarter. The results to date are as follows.

The response of the matrix to the true step in fluid temperature can be represented by the formula

$$t-t_i = (t_1-t_i)\ \nu(\tau,z) \tag{3}$$

where t is the gas temperature at the dimensionless position z and time τ, t_1 and t_i are the corresponding initial fluid and solid temperatures. (A similar formula can be written for the solid temperature but will not be needed here.) In view of the linearity of the basic equations, the outlet gas temperature for an arbitrary inlet temperature variation can be written as a Duhamel integral (2). For the infinite conductivity matrix, the analysis can be carried out and the result expressed in closed form. From Mondt's analysis (1) for the function $\nu(\tau,z)$ evaluated at z = 1, the matrix outlet, we have

$$\nu(\tau,1) = 1- Ntu\ \psi e^{-\tau\psi} \tag{4}$$

where $\psi = (1-e^{-Ntu})/Ntu$. Defining a matrix time constant by the formula $\theta_m = \bar{C}_s/\bar{h}A\psi$ and performing the analysis referred to above gives, for the outlet gas temperature, the expression

$$\frac{t - t_i}{t_1 - t_i} = 1-e^{-\theta/\theta_s} + \frac{\psi Ntu}{1-\theta_s/\theta_m}\left(e^{-\theta/\theta_s}-e^{-\theta/\theta_m}\right) \tag{5}$$

It has not been possible as yet to evaluate the integrals for the zero conductivity case for which an exact formula for $v(\tau,z)$ is also available, nor have the implications of Equation 5 been fully explored.

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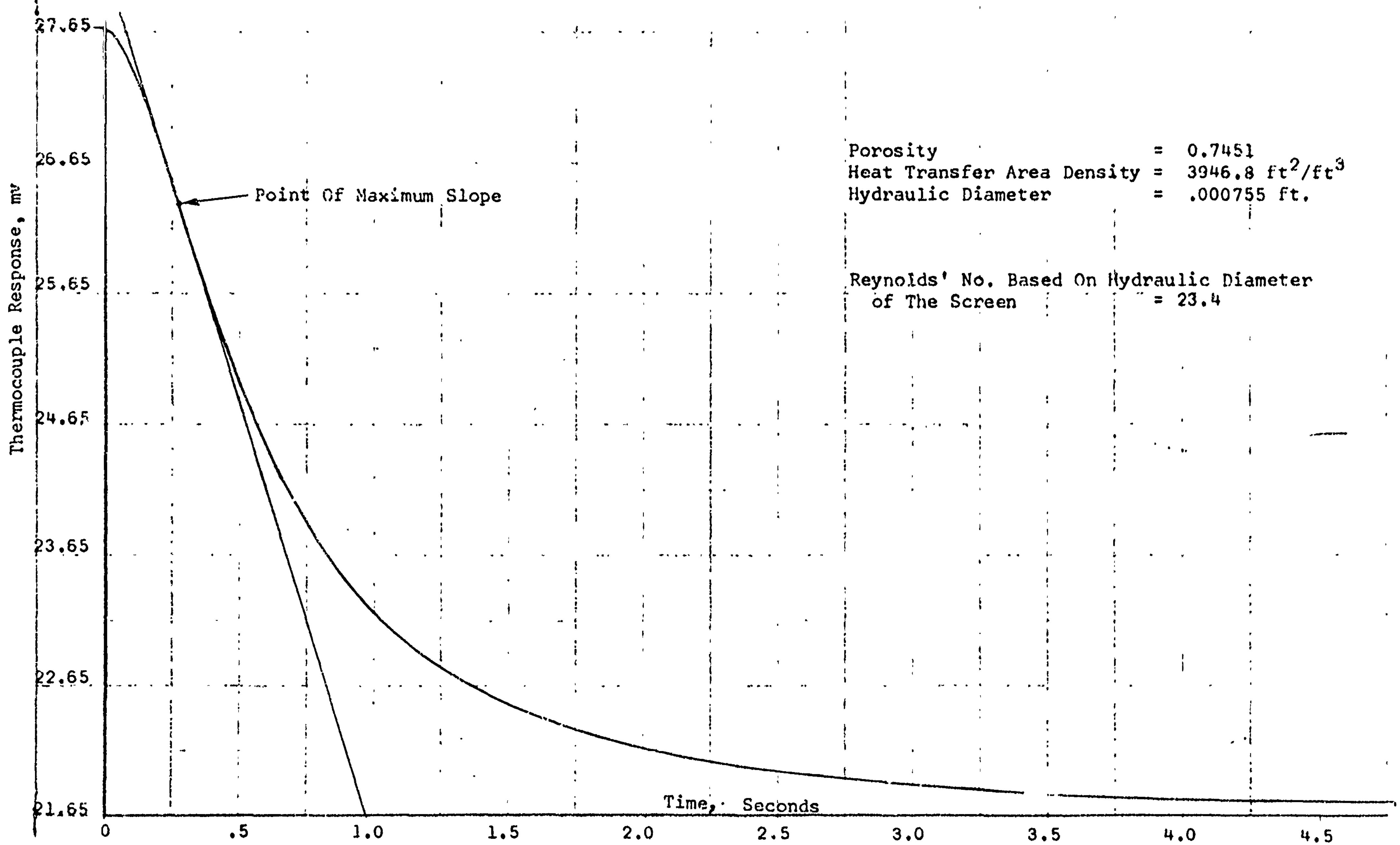
Figure 2 COOLING CURVE OF 0.0031" WIRE DIAMETER, 100 MESH, NICROME V SCREEN



Appendix B3

FLOW VISUALIZATION

This experimental facility has been designed at Intertech and is located there and will provide information in a range of scales from 3 times to about 16 times actual size. A simple low-velocity air duct has been assembled with flow created with various sizes of standard room fans. Air velocities in the test section will be between 1 to 10 ft/sec.

Firstly, flows will be visually studied: 1) with optical means using parallel beam Schlieren and shadowgraph systems with suspended particles, (aluminum, kapok) released into flow at inlet, and 2) with standard smoke techniques and fine silk threads, and 3) with china clay and titanium tetrachloride oil evaporation techniques. Also, other specialized methods will be used to investigate boundary layers, air velocity profiles and diffusion rate through the plates. Various hole sizes and slots will be tested as well as actual matrix plates cut to fit the test box. It is hoped that these simple tests will shed light on the flow characteristics associated with flow through a plate matrix. Effects of parameter changes such as hole size, relative pattern, plate thickness and passage height will be clarified.

Secondly, heat transfer will be investigated from one or more heated plates into the stream of air. By means of sandwiching nichrome nets, plates can be manufactured that will be heated to desired temperature. Due to characteristics of plexiglas, only relatively low temperature potentials will be used, say 30 to 50°F. It is felt that, although qualitative results might be difficult to obtain, local areas of effective heat transfer can be established. If sufficient measurements can be a basis for the heat transfer efficiency of the plate surface can be established and thus various configurations compared.

It is a necessary but not sufficient condition for flow similarity to produce equal Reynolds' numbers based on average velocity and size of one passage (hydraulic diameter). The turbulence level will remain unknown in these tests because at present no hot wire equipment is available at Intertech. The air velocities in passages are low, and thus the pressure drop through the model matrix is too small for accurate measurement by conventional means. However, designs have been made for a very sensitive manometer which would be capable of measuring pressure differentials of less than 0.001 in.H_2O. With this instrument, standard pitot-static tubes could be used at very low speeds.

The friction factor of a given perforated or slotted plate is compared to a smooth plate under identical conditions of air velocity and temperature. Increase in friction factor is likely to happen with increasing film coefficient. It is felt that the pressure drop through the matrix can be more accurately investigated in matrices of actual size. This flow visualization setup would only provide comparative information for various plate configurations.

The flow (cfm) passed through the 4-inch square test section will be reduced by a factor of scale factor squared. Two passage sizes will first be investigated, 0.5 in. and 1.0 in., both with 0.25 in. thick plates. The first case will allow 5 passages; the second only 2. Various hole sizes and slots will be tested by interchanging the internal plates in the test section. It is realized that scale-model testing has its shortcomings, and in this case it is likely to be difficult to evaluate in concrete terms, but any information on characteristics of flow over the various plates will be valuable.

Appendix B4

TURBULENCE GENERATOR

Rotating Pinwheel Turbulence Generator

The purpose of the pinwheel is artificial promotion of higher turbulence levels in flow arriving at the test section.

The design that will be made incorporates a ring which will be inserted in the round portion of the tunnel after the inlet cone. The ring will have the same internal diameter as the tunnel at that point and will be supported by three rubber tired wheels so that it is free to rotate. The drive will be through one of the wheels with provision for driving the ring at different speeds.

Provision will be made for attaching several rods to the inside of this ring such that they project inward toward the center but do not necessarily touch. The resulting ring and rods is called a pinwheel.

The initial tests will be done with six cylindrical pins 0.75 inches in diameter and three inches long. These pins will create eddy wakes of comparatively known character. The decay, for example, in a given flow can be approximately predicted. It is felt that these wakes simulate closely those existing after the last stage of actual turbomachinery. The turbulence level can be characterized by measurements using hot wire or total pressure methods of mean motion and fluctuations in the air stream just ahead of the testing section.

The pinwheel will cause a certain flow resistance and pressure drop, especially with a large number of pins. This would necessitate pressure data in space between pinwheel and test matrix in addition to turbulence measurements. It is quite possible that various perforated matrices will react differently to high levels of turbulence. Thus after installation of pinwheel several combinations should be tried out for their heat transfer characteristics.

Turbulence Grids

Informative data have been published by NASA, NACA and others[1,2,4] on measurements made by well-established techniques to give the influence of the screen mesh size on the turbulent intensity, scale, and the power-spectral-density. The results show a linear dependence of the intensity upon the screen mesh size for locations within the central core of the air jet. The flow becomes turbulent after a screen is inserted therein, because eddies are shed from the wires forming the mesh.

The initial size and intensity of the eddies are determined by the dimensions of the screen. Very close to the screen the wakes from the individual wires influence the turbulence, but this effect soon disappears within approximately 15 mesh lengths, leaving the turbulence uniformly distributed. Since the turbulent motion decays rapidly, the intensity of turbulence decreases with distance from the screens.

Matrices of crossed-rods to simulate woven screens are used to create higher intensities of turbulence. Reynolds' Numbers of 5 to 1000 and porosities of 0.60 to 0.83 have been investigated and data published.[3]

Final design for six grids are now being made and at least two will be fabricated for initial testing. Experimental studies will have to be made to determine the location of the grids relative to the core and the effect of grid material on the temperature measurements during heat transfer tests.

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